THE DESIGN OF THE BART-AFC BARRIER DRIVE SYSTEM AT IBM (A)

This case describes in part the design effort and decision process associated with the development of an actuating system at IBM, San Jose. The actuating system powers a pair of barrier leaves in the automatic gates of Bay Area Rapid Transit District Stations. The case starts when delivery is due in 16 months and only an unsatisfactory prototype exists.

Part A describes the studies of competing electric and pneumatic solutions and the selection of pneumatics. Part B follows the design of the leaf and actuating system from the contract specifications through a functional schematic, sizing calculations, the availability of appropriate commercial components, the competitive testing of these components and their final selection. Part C takes a similar look at the development of the compressor system.

 Prepared by Prof. R. Piziali with support by the San Jose Division of IBM Corporation. Assistance by C. Donelson, W. Goddard, J. Minick, G. Pastre, and K. Sanders is gratefully acknowledged.

THE DESIGN OF THE BART-AFC BARRIER DRIVE SYSTEM AT IBM (A)

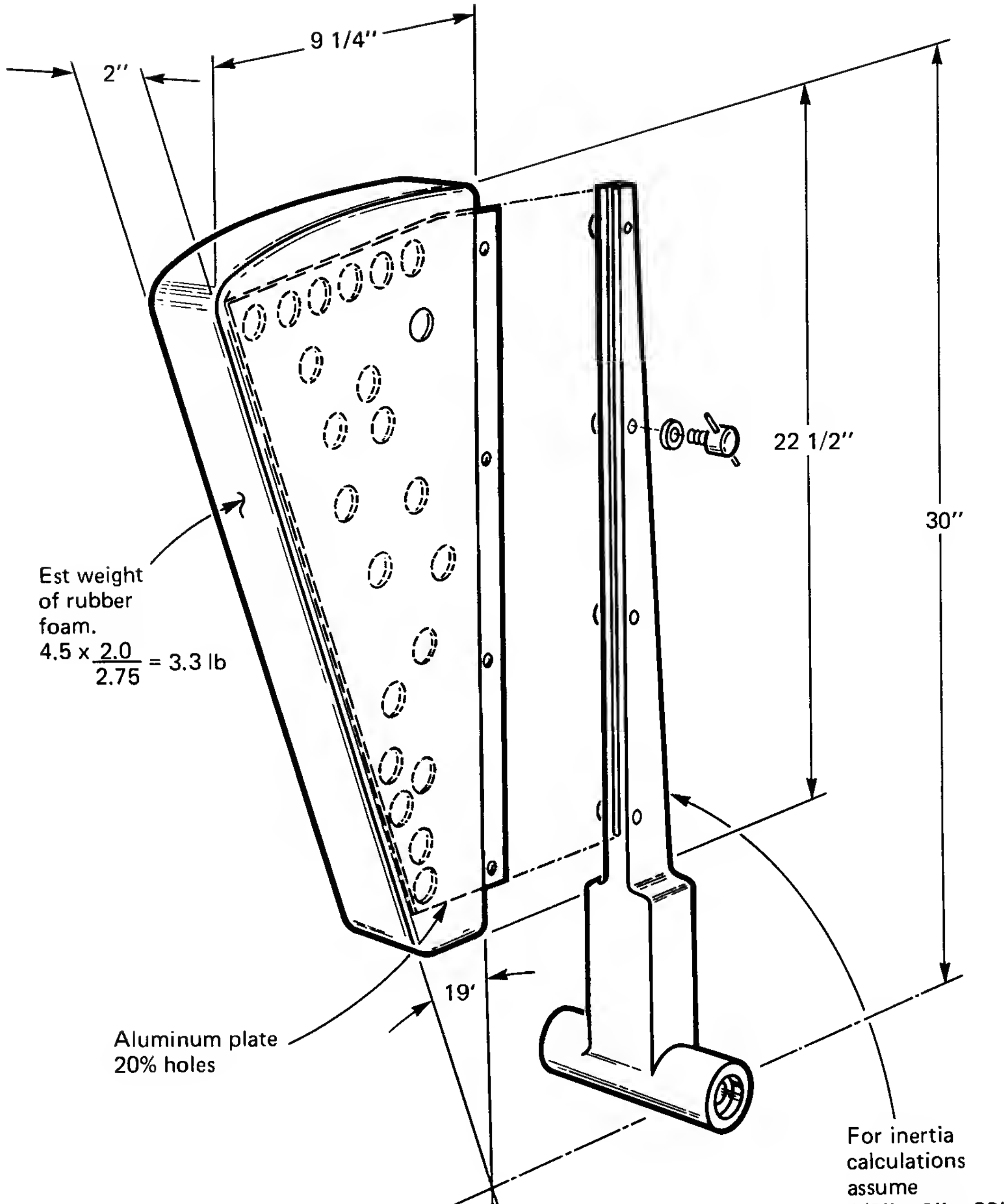
Selection of an Actuating System

From its inception the Bay Area Rapid Transit (BART) commission was convinced that a modern, convenient transportation system was required in order to assure wide acceptance in the San Francisco Bay Area. An integral part of their concept was the automatic fare collection system (AFC). They invisioned a typical passenger entering a station, going to an automatic ticket issuing machine and purchasing a magnetically coded ticket. The passenger would then deposit this ticket into a gate, the gate would magnetically indicate the entrance station, return the ticket, and allow the passenger to pass through the barriers. At his destination, the passenger would deposit his ticket into an exit gate. This gate would subtract the value of the ride from the value of the ticket, return the ticket and allow the passenger to exit through the barrier. The BART commission issued a 123-page contract describing in detail the functioning of the AFC System along with specific options, service capabilities, reliability tests, and installation requirements. However, the design of the components to satisfy these specifications was left entirely to the contract bidders. A station attendant would be available only to oversee station operations and therefore, the bidders had to consider the design of various change making, ticket issuing, and ticket reading equipment as well as automatically operating gates. At the conclusion of the bidding, the IBM Corporation was awarded the AFC contract.

It was on May 1, 1970 when Steve Campbell, a project manager in SDD (Systems Development Division), was assigned the responsibility of developing the mechanical engineering aspects of the AFC system at IBM San Jose. Steve had been with IBM for twelve years after receiving his Masters degree in Mechanical Engineering and was familiar with

developing equipment within general specifications. He knew that specifications were written to describe a desired function and do not concern themselves with detail design. Therefore, contract specifications sometimes exist which cannot be designed into reasonable engineering hardware. Consequently, contract renegotiations occur in order to develop specifications that comply with hardware capability. One specification, however, that could not be negotiated was that of delivery time. The design of the system was presently in the early prototype state, and component delivery and installation in BART's 33 stations was to begin on September 1, 1971, just 16 months away. Such a schedule would require drawing releases by mid September 1970 and normal development procedures would not be sufficient. Steve knew he would have to draw heavily on the engineering expertise of IBM in pursuing parallel and overlapping paths in order to develop a quality product within this time constraint.

Steve was responsible for the development of the ten major components which go into a typical BART station. He divided his people into two groups, each headed by a project engineer. One group was to develop the gates and ticket readers while the second group worked on money changers, addfare units and ticket vendors. Within these groups lead engineers were assigned areas of responsibility, e.g., a lead gate engineer. One of the design problems associated with the gates was the development of a barrier leaf and drive system. An early barrier leaf design and barrier operation schematic are shown in Figures 1 and 2. This barrier leaf is foam padded for passenger protection and constructed for minimum weight within its strength requirements. A complete barrier consists of two such leaves, each mounted on one of the two gates which forms an entrance or exit aisle. The barrier drive system had to open

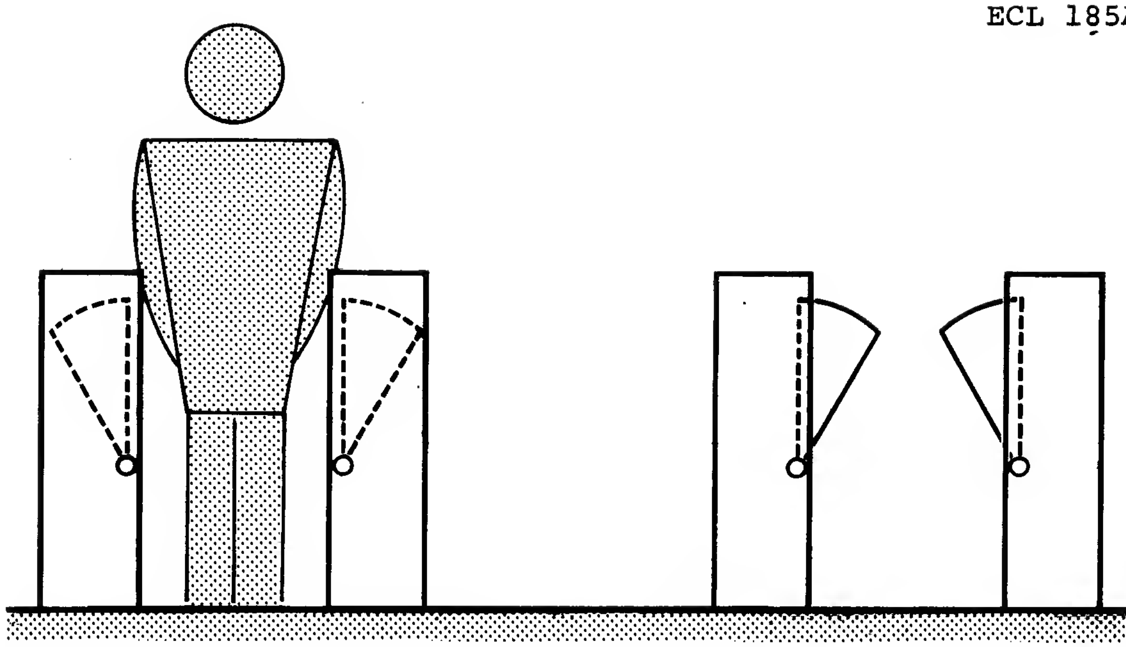


BART Gate

Provisional - for inertia calculations

All dimensions are approximate

Fig. 1 Early barrier leaf design.



Barrier: gate open Barrier: gate closed

Fig. 2 Schematic of gate operation.

(or close) the barrier in .4 ± .1 second while maintaining a stall force of 10-30 lbs (adjustable).

Other specifications (Exhibit A-1) included a maximum load of 40 passengers per minute, an alarm system, an adjustable stall force, the ability to open and latch during power failure, and detail size and strength requirements. The prototype gate design indicated that the size of the actuating system would also have to be small, but specific size constraints had not yet been established. Some general gate requirements also influenced barrier system design. These included modular construction, a maximum modular weight, and serviceability of a gate from its disabled aisle. Since each gate was part of two aisles, this last requirement insured that one aisle failure would not effect other aisles.

The prototype barrier system had been developed using pneumatic power, but this system was not meeting specifications and there was

some question concerning the reliability of compressors. As a matter of fact, one week before SDD Engineering took over, the Advanced Systems Development group (ASD) at IBM Los Gatos was asked to head a study of pneumatic versus electrical actuators.

This study was to determine the cost, serviceability, reliability, availability and packaging of four power sources; pneumatic cylinders, rotary electric motors, a special solenoid, and linear electric motors. The ASD group was required to transform the contract performance specifications into component specifications and then to locate available components in order to estimate the characteristics of each power system. However, they were not expected to do a detailed design and therefore, the actual assembly configuration, alarm systems and other details were not considered. Using six full-time and four part-time engineers, this project was completed in two weeks. Its results (summarized in Table 1) show a preference for the pneumatic system, particularly since development time was such a critical factor and serviceability could be improved by good packaging design.

Concurrent with the ASD study was a quick look at pneumatic actuators and a collection of all IBM test data on compressors by two engineers from IBM's Technology and Advanced Development group. This study noted that the prototype high pressure compressor (175 psi) would not meet the specified life requirements. This compressor had been selected to achieve a restraining force of 30 lbs. This resulted in considerably more torque than was required to open the gates in .4 second. They concluded that the restraining and opening functions should be accomplished by separate systems, that the pneumatic system should be redesigned and that electrical alternatives should be examined as possible replacements.

Table 1 ASDD barrier drive study summary

	Air	Motor	Solenoid	Linear Motor (not yet available)
Reliability	Good	High	High	Very high
Serviceability	Poor (present design)	Good	Good	Good
Power requirements	Low	High	Very high	High
Space requirement factor	1.0	1.2	1.5	1.0
Development time	Low	Longer	Longer	Longest
Cost per gate	$237	$682	$715	$200

After seeing these reports, Steve felt that pneumatics were potentially the best solution but that a detailed electromechanical design should be developed concurrent with pneumatic redesign. He assigned the pneumatic redesign to Dave Bingham, an experienced designer in the BART-AFC group, and he assigned the electromechanical design to an independent IBM Task Force of four engineers. These parallel efforts were to consider space requirements as well as the previous considerations. Although minor modifications were possible, the barrier system, including two barrier leaves, their actuators, controls, and alarm had to be located in a space 11" x 11" x 38".

The Task Force started by reviewing the ASD study. They rejected the use of linear actuators as a noncapable design concept and felt that the rotary motors suggested by ASD would not be practical for barrier actuation. They noted that the barrier, with an inertia of 4 in-lbs-sec and a maximum gravity torque of 11.2 in-lbs could be properly actuated by 45 in-lbs torque. However, in order to resist forcible entry at 30 lbs a resistance torque of 600 in-lbs was required. They solved this dichotomy by selecting a motor-brake system. The motor was sized for operating torque, the brake for resistance torque, and limit switches were used for control (Figure 3).

With the performance characteristics of a standard commercial motor (rated at 30 in-oz) and a 30-1 gear reduction, the group ran a computer simulation of barrier actuation. They included all system effects except friction and braking torque. The resulting operation time of .285 seconds led them to believe that friction and braking would not increase the operating time beyond .4 ± .1 second. On June 4, 1970 the Task Force submitted their report to Steve Campbell, concluding that the design could be built for approximately $150, would consume only 71.5 watts, and would have high reliability. Problem areas would include motor temperature rise, gear wear and efficiency, and crash stop design for long life. They recommended that an engineering model be built and tested.

Concurrent with the electrical motor development was Dave Bingham's redesign of the pneumatic system. He first examined the prototype system. It used a 1-inch diameter cylinder operating at 40 psi air pressure. He found that its operation was too slow to meet specifications, that it was difficult to service, and that the rubber cushion stops failed quickly. He then decided that the best course of action was a

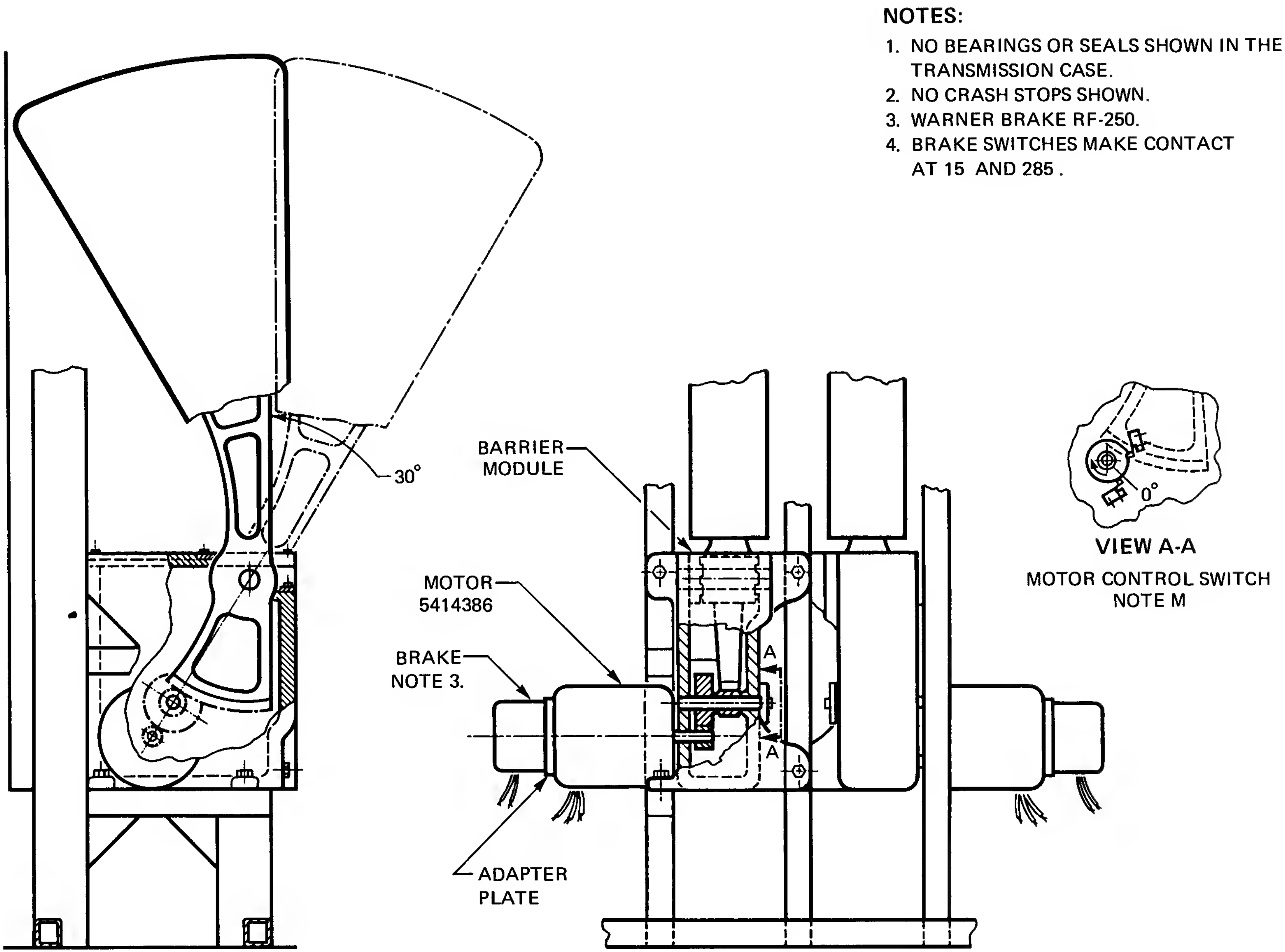


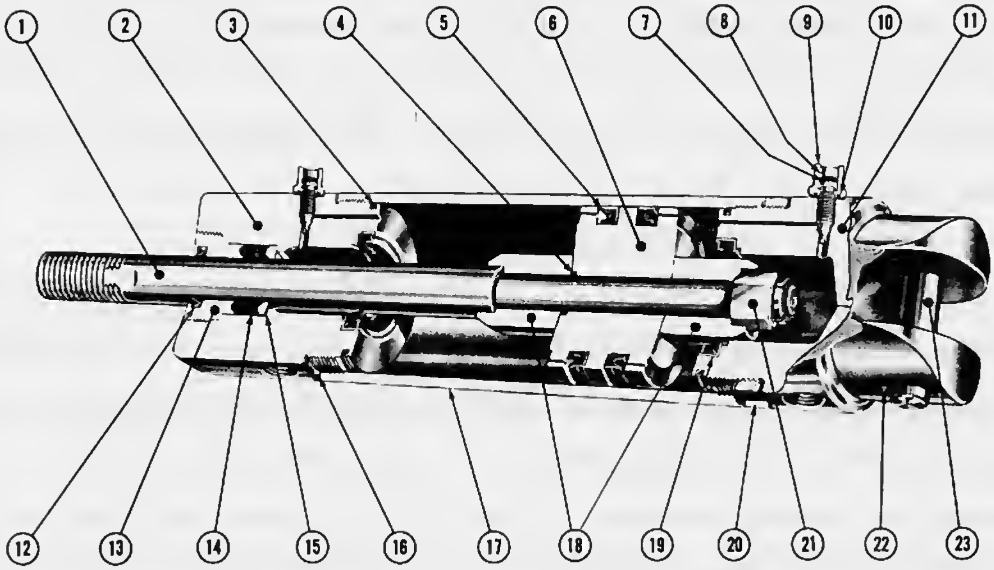
Fig. 3 Electric motor-brake actuator design.

complete redesign starting with the barrier leaves themselves. The 30-inch leaf (Figure 1) had been overdesigned relative to its minimum area specification of 75 sq. inches and Dave was able to reduce inertial forces by designing an 18-inch barrier. This barrier was used in the Task Force design and is shown in Figure 3.

Dave's next step was to determine the parameters of a pneumatic cylinder capable of operating this new barrier within the specified time. The calculations were complicated somewhat by the fact that both the gravity force and the tangential force component of the cylinder on the barrier varied with position. Using detailed inertia calculations in a computer simulation, Dave was able to determine barrier operation times for various pivot points, air cylinder sizes and operating pressures. He found that a 1 1/2" diameter cylinder with a stroke of 2.8 inches and an operating pressure of 40 psi would open the barrier in .103 second. Realizing that the solenoid valve time delay, the line delay and frictional forces would increase the deployment time, he decided to use the above parameters for pneumatic cylinder specifications. Additionally, he specified that the cylinder have a built-in air cushion. This would decelerate the barrier leaves and therefore extend the life of the rubber stops.

With specifications established, Dave searched catalogues to find an appropriate cylinder. He commented that it was normally quicker and cheaper to select very good components for prototype testing. This ensures a minimum of failures and a better isolation of problem areas. Once the system is working and understood, cheaper components can be tested for possible substitution. Dave selected an Air Products model GG-200 air cylinder (Figure 4). This cylinder was 10 inches long, rated

FOR AIR . . . TO 200 P.S.I.



PARTS LIST FOR BORE SIZES 1½" THRU 3"

KEY NO.	PART NAME		NO. REQ'D
1	ROD—PISTON (Specify cyl. stroke)	—Non-cushioned —Cush. rod end —Cush. blind end —Cush. both ends	1 1 1 1
2	HEAD—ROD	—Non-cushioned —Cushioned	1 1
3	"O" RING—BODY SEAL		2
4	"O" RING —PISTON TO ROD SEAL		1
5	PACKING—PISTON		2
6	PISTON		1
7	"O" RING CUSH. ADJ. SCR.	—Cush. rod end —Cush. blind end —Cush. both end	1 1 2
8	SCREW—CUSH. ADJ.	—Cush. rod end —Cush. blind end —Cush. both ends	1 1 2
9	LOCKNUT—CUSH. ADJ.	—Cush. rod end —Cush. blind end —Cush. both ends	1 1 2
10	"O" RING CUSH. ADJ. LOCKNUT	—Cush. rod end —Cush. blind end —Cush. both ends	1 1 2
11	HEAD—BLIND	—Non-cushioned —Cushioned	1 1
12	SCRAPER—ROD		1
13	NUT—PACKING		1
14	PACKING—ROD		3
15	ADAPTOR—PACKING		1
16	LOCK WASHER—CYLINDER		2
17	BODY (specify cyl. stroke)		1
18	COLLAR—CUSHION	—Cush. rod end —Cush. blind end —Cush. both ends	1 1 2
19	SEAL—CUSHION	—Cush. rod end —Cush. blind end —Cush. both ends	1 1 2
20	LOCKNUT—CYLINDER		1
21	LOCKNUT—PISTON ROD		1
22	COTTER PIN		2
23	PIN—SWIVEL		1

Fig. 4 Air Products model 66-200 air cylinder.

at 200 psi and cushioned on both ends. The air cushion arrangement can be seen in the figure. During the first part of the stroke the exhaust air passes unrestricted to the exhaust port. However, near the end of the stroke, the cushion collar (Figure 4, No. 18) reaches the cushion seal (No. 19) and air is forced to exhaust through a controlled (No. 8) side passage. The restricted flow in this passage increases exhaust chamber pressure and reduces piston velocity.

In the early days of the BART-AFC group, there were many personnel changes, and at this point Dave was assigned to another area of the project. The barrier design responsibility was assigned to Tom McDowell, who was also an experienced IBM designer. Even though they worked together for a short time, Tom repeated most of Dave's work in order to attain sufficient confidence and background in the design. Tom started with the barrier leaf, reducing its size slightly and changing from a one-piece aluminum casting to a two-section barrier leaf. The redesign resulted in reduced cost and increased strength. Tom then decided that a 20 psi air supply would be sufficient for barrier actuation. It would also be more efficient, using less air per actuation and reducing leakage losses. He had to assume that the required stall force of 30 lbs. could be renegotiated to a lower value. In addition, Tom included low friction to his list of cylinder specifications. Excessive friction could cause nonsimultaneous operation of the two aisle leaves. This would result in a sloppy and unacceptable performance. The high break-away friction and added high cost of the Air Products cylinder led Tom to look for alternative cylinders. From his own files and those of IBM's purchasing department he was able to contact many pneumatic cylinder manufacturers. His search led him to select a Control Air pneumatic cylinder because of its low breakaway friction, low cost and acceptable life characteristics.

At this point, further BART group reorganization provided Tom with a new supervisor. Bob Everest came to the BART group as lead Mechanical Engineer for the entire gate system. Bob had been with IBM for twelve years as a mechanical engineer and had 25 years of engineering experience. He had worked on paper tape and card punch machines and more recently on an electron beam technique for writing on film. It was near the middle of June when Bob started on the AFC project and the target date for selecting the barrier drive system was set for June 22.

Bob and Tom decided to have one more look at their options. They both favored a pneumatic drive, but wanted to be sure that no workable alternatives had been overlooked. Tom had reviewed the ASD study and the Task Force design. Actually, he and Dave Bingham had rejected the Task Force design when they were working together. First, the gear box was subject to wear, maintenance, and lubricant leakage problems; second, the automatic open during a power failure would be difficult to implement; and most significantly, the system was occupying restricted space. Bob and Tom decided that the only electrical components with design potential were linear actuators. Tom looked in detail at three possibilities. An actuator was available which converted rotary motion to linear motion with a ball screw assembly. By looking through the company's literature and talking with their sales representative, Tom found that the unit needed to meet the force specifications would cost $250, and have a 4.75-inch square body and a 10-inch long shaft with 5 inches clearance required for a ball nut and rear shaft allowance. This was longer than the allowable actuator space. Also, its excessive stall force could injure a passenger caught in a gate, and, although its power consumption was only 90 watts, its 3-phase 220-volt power requirement would necessitate a

transformer. The actuator company was interested in getting the IBM contract and proposed manufacturing a specially designed unit. This, of course, would require development time and life testing, so Tom and Bob decided against it.

Another linear actuator was essentially a flat, linear squirrel cage motor. Literature and sales representatives indicated that the best unit would be small enough and cost only $130. However, its power consumption was excessive (548 watts at full load); its available force was marginal (10-12 lbs); and there were certain mounting and control problems. Tom also considered an actuator which converts the rotary motion of a shaft to the linear motion of a small drive unit. With three free-wheeling rollers on each end of the drive unit, angled to describe a helical path along the shaft, the drive unit moves along the shaft according to the direction of rotation. A complete system then requires an electric motor, shaft support bearings and the actuator, making it a complicated system and probably impossible to package.

Two in-house designs using an a-c torque motor with either a ball screw drive or direct gear drive were also rejected because of extensive development time, large commercial components, and performance uncertainties.

Tom decided that while the electric actuators had potential as design solutions their development time would be excessive within the project time constraints. He felt that the time constraint forced him to stay with established technology and to use only proven components. Bob voiced a similar opinion when he stated that he favored pneumatics

because he had seen them work in many similar applications over the years and he knew they would do the job. The linear electric actuators presented a newer technology that would require excessive development time.

Therefore, in less than two months, the BART-AFC barrier design had involved 19 IBM engineers and designers who examined 14 possible electromechanical drive configurations and redesigned the pneumatic drive system. They had reached the point where a firm commitment to a drive system was essential. Throughout a period of product development of this type the interaction of different levels of management occurs frequently. The designer, lead engineer, project engineer and project manager are all advised of investigation and test results and the steps taken to meet existing schedules. In this case all of the above people had been following the search for a drive system and all preferred pneumatics. Steve Campbell asked Bob, as lead gate engineer, to write a memo summarizing his findings and supporting the selection of pneumatics. Steve then presented Bob's memo (Exhibit A-2), along with his endorsement, to the Program Manager. Although the selection of pneumatics did not find easy acceptance at the next two levels of management, by the end of June they were firmly established as the power source of the barrier system.

<u>Passenger Gate (Barrier) Specifications</u> (Taken from BART contract specifications)

1. Processing time of entry and exit gates, measured from the instant a patron inserts a ticket until the ticket is returned, or until the barrier is released if a ticket is to be captured, shall not exceed 0.3 second. Gates and their control system shall be designed to maintain a rate of not less than 40 patrons per minute, including the possibility of all gates in the system being activated simultaneously.

2. The operation of the barrier shall be such that a patron may stall the closing motion of the leaves with a nominal force of 20 pounds per leaf at any time throughout its stroke, including full closed. This force shall be adjustable from 10 to 30 pounds continuously or in increments not greater than 5 pounds. This force shall be applied at the point of maximum travel of the leading edge. The leaves shall move through their full stroke in not more than 0.4 second plus or minus 0.1 second.

3. Upon complete loss of power to a gate, its barrier shall be openable, and shall latch open, by a person applying a force on the leaves not greater than the stalling force indicated. The latch shall release automatically when power is restored, to allow the control of the barrier leaves to return to the normal mode of operation. Provisions shall be made for locking the barrier closed, by using the maintenance key.

4. When the barrier leaves are fully closed, an alarm circuit shall be activated such that any attempt to force the leaves open shall cause a vibrating bell to sound in the gate console. The sound pressure level of this alarm bell shall register at least 100 db on the C-scale of a sound pressure meter 10 feet from the console.

5. The control of the leaves shall include the detection of patrons in the gate passageway. The design shall preclude the closing of the barrier upon a patron, or upon packages which the patron may be carrying in front or behind him, but minimizing the possibility of two or more patrons traversing the gate upon the payment of only one fare.

EXHIBIT A-1

6. The barrier leaves, when closed, shall present a minimum surface area to the patron's direction of travel of 75 square inches per leaf. These surfaces shall be padded. The padding shall conform to FS ZZ-C-00811b, Class 2, 3/8-inch thick rubber cushion, or as approved by the Engineer.

7. The leading edges of the leaves shall be covered with protective padding of suitable thickness and hardness to protect patrons from harm in case of contact.

8. When closed, the top of the barrier shall be within five inches of the top of the console, but at no time shall the barrier extend above the top of the console.

9. Impacts upon a closed gate, in both directions of travel, equivalent to a 200-pound man moving at three miles per hour (4.4 ft./sec.) striking the barrier at the centerline of the passageway, shall cause no permanent deformation or other damage, and 500 such impacts shall cause no discernible wear or fatigue on any part of the console and its mechanism. In addition, a 300-pound downward force on the barrier shall cause no permanent deformation or discernible damage.

ECL 185A

IBM

Date: June 22, 1970

From (location or U.S. mail address): FTS Mechanical Development

Dept. & Bldg: H26/060

Tieline & Tel. Ext.: 1373

Subject: Barrier Operation; Air vs. Electric

Reference:

To: Project Manager

The barrier must complete its opening or closing actuation within 0.4 (±.1) seconds after the starting signal has been given. In addition, the barrier must not exert a force greater than 10 to 30 pounds against the customer. The minimum force is sufficient to actuate the system, and with this assumption, the attached comparison chart has been made.

In the comparison, the air system was evidently the more desirable. In addition, the successful operation of the existing system compared with starting the development of a new system also directs our decision to using the air system.

Note: The attached detail information represents a more complete analysis of the electrical components, as well as comments on the use of pneumatic actuators.

Bob

EXHIBIT A-2

Barrier Operation Air vs. Electric Comparison Chart

	AIR	ELECTRIC
Adjustment	Adjustment very simple and flexible through pressure regulation.	Requires voltage control and/or some logic control.
Space	Cylinder and components fit into area provided.	Unit is larger and will require considerable effort to fit into allotted space.-- Also - larger transformers.
Power	Power source external to gate and allows flexibility by choice of compressor.	The Unit, which was most promising, requires a 220V, 3 Phase power, which is not present at the gate. This results in a special unit which requires extra testing, is not stocked, and is higher in cost.
Stopping Characteristics	Standard air cushions at both ends of the cylinder provide minimum shock at stopping.	A motor brake is required which is subject to wear and, in addition, limit switches must be used to sense end of travel.
Power Failure	Electric power failure sets valves to open gates, which is considered very desirable to the BART people.	Stops under power failure and gates must be pushed out of the way with a force dependent on type of gear or worm used.
Environmental Conditions	Through the use of appropriate materials, air cylinders can be used in hazardous environments without undue cost.	Motors capable of being used in hazardous environments are usually sealed and are considerably more expensive.

EXHIBIT A-2

ECL 185B

ENGINEERING CASE LIBRARY

THE DESIGN OF THE BART-AFC BARRIER DRIVE SYSTEM AT IBM (B)

THE DESIGN OF THE BART-AFC BARRIER DRIVE SYSTEM AT IBM (B)

Development of the Actuating System

The decision to develop a pneumatic barrier system was firm by July 1, 1970 and the target date for drawing release was still September 1970. This established two patterns of operation. First, testing and design would have to take place concurrently. And, second, drawings would be released and updated with the firmness of their status noted (i.e., firm, possible review or definite review). While the AFC group felt that they must release the drawings in order to develop lead time, the drawing status indicated whether purchasing should place an order for the total number of parts required or for a smaller amount to use in prototype construction and manufacturing set up. It was also imperative to maintain close cooperation between the engineering and manufacturing groups. More and more the manufacturing engineers are consulted in the early stages of design to give advice regarding manufacturing processes and assembly procedures. Engineering also knew that they could call on their technical services group for information on materials, finishes, and lubrication.

Because of their commitment to established technology, the design approach was principally one of selecting, interfacing, and packaging off-the-shelf, reliable components. However, before these components could be sized, the question of stall force had to be resolved. The establishment of a 30-lb resistance force during forcible entry had always been a design problem. Bob and Tom felt that this was not a reasonable specification. A force of 30 lbs was too low to prevent forcible entry and therefore the primary mode of forcible entry prevention was the alarm system. Steve was in agreement with them and as program manager, set the proper renegotiation machinery into motion. The suggested variation

was agreed to by the BART negotiators and the maximum resistance force was reduced to 10 lbs. Recalculations (Exhibit B-1) ensured proper cylinder performance at 21.2 psi.

As work began to pile up, Bob assigned the barrier actuator and a alarm systems to Tom while he accepted the compressor system as well as his overall gate responsibilities. A schematic/showing the components and function of Tom's pneumatic design is shown in Exhibit B-2. There are two variations between this design and that of the prototype. The prototype also used two 3-way solenoid valves but arranged with one power valve and one exhaust valve. Tom decided to use one 3-way valve for opening power and one 3-way valve for closing power. This required two mufflers instead of one but reduced the number of fittings and resulted in simpler pneumatic circuitry. As the schematic shows, Tom had to select seven components: an air cylinder, a 3-way solenoid valve, a pressure regulator, a filter and automatic drain, an alarm system, and pneumatic fittings and tubing. In order for a component to be released for purchase, it was required to satisfy performance requirements and IBM safety requirements, to attain the approval of Underwriter's Laboratories and to pass contract reliability tests.

Tom followed the normal pattern of searching through catalogs to find possible sources and then checking with manufacturers and salesmen on components which looked promising. He had already selected two air cylinders and decided that these would be sufficient. His selection of a 3-way solenoid valve also involved two manufacturers, QVI and Valvco. Tom favored the QVI valve because its larger orifice created less flow resistance and it could be purchased with a high temperature coil. Valvco offered to produce a specially designed valve for IBM, but envisioning development delays, Tom decided to test only the QVI valve.

In searching for other components, Tom contacted a salesman from Pneumatics, Incorporated (P.I.), a local distributor of pneumatic components. Because P.I. carried a good line of components and because their sales representative was knowledgeable and helpful, Tom found himself selecting minor components from the P.I. catalog. He decided to use the Success type 317 miniature filter and the Success type G31 compact pressure regulator. Tom did not really like the Air Products cylinder and the P.I. salesman convinced him to test an Emerson model 12 as a third possible choice. In selecting pneumatic tubing, Tom faced the dilemna of requiring tubing large enough to insure proper actuation times and small enough to package in the allotted area. This led him to the selection of 1/4" o.d. - 3/16" i.d. nylon pneumatic tubing. He wanted a plastic tubing for ease of installation, but soft plastic tubing required large fitting inserts which greatly restricted air flow. The nylon tubing was soft enough to bend, yet hard enough to use standard metal fittings. He later established that actuation times would have been excessive if a 1/4" soft tubing had been used.

As the alarm system appeared straightforward enough, Tom looked until he found a unit of reasonable performance and price. He selected a PUI model 120 vibration horn and had it tested to ensure that the required noise level specifications were met. When the horn passed these tests, it was released for purchase.

While he was selecting components, Tom stayed in close contact with IBM's Product Test Laboratory. Product Test is independent of the development laboratory (SDD) and therefore not subject to the pressures of testing and accepting their own design. This group normally performs

four series of tests. They follow a development test on a conceptualized system with more detailed tests on a custom built model. Their third series tests a manufacturing prototype to allow the start of production. The final tests are made on actual production systems before release to customers.

In the case of the BART-AFC system, normal procedures were again abandoned. Not only was there extensive interaction between test and engineering, but the test group for the first time was subject to difficult time constraints. Tom worked with the test group to develop complete actuation systems for performance and reliability. As the barrier leaves were not yet built, a dynamic mock up was constructed and the system was powered by the building air supply. These tests would run for 1.5 million cycles and components would be examined at that time unless failure had occurred earlier. A tentative schedule proposed that component evaluation tests start on September 7, 1970 and that they be completed by November 9, 1970. However, due to various delays, the first cylinders actually began testing on September 21, 1970.

As the testing of the pneumatic components proceeded, Tom turned his attention to the detail design of the barrier leaf and packaging. As the design had to be developed before the tests results were available, Tom assumed that the Control Air cylinder would be used and that all other components would function properly. For the barrier leaf, he had decided to use a two-piece design (Exhibit B-3) so that the molded section could be quickly and simply replaced if the rubber foam became worn or damaged. His stress calculations were based on the BART specifications (Exhibit A-1) which stated that "impact upon a closed gate in both

directions of travel equivalent to a 200-lb man moving at 3 miles per hour striking the barrier at the center line of the passage way shall cause no permanent deformation or other damage and 500 such impacts shall cause no discernible wear or fatigue on any part of the console and its mechanism. In addition, a 300-lb downward force on the barrier should cause no permanent deformation or discernible damage." As it turned out, a leaf designed to satisfy the former requirement easily satisfied the latter.

For the mounting shaft and arm (Exhibit B-3) these calculations determined basic dimensions while for the barrier they also determined the mode of construction. Tom favored using an aluminum slab design because it could be cut from a stock sheet and construction costs would be minimum. However, if strength calculations required the slab thickness to be physically large a cast aluminum or welded sheet metal box construction could be used. Unfortunately, Tom no longer has record of his calculations, but he recalls making the assumption of constant deceleration over an assumed distance. He divided the 200-lb body at the trunk and considered the deformation of the body and the barrier leaf cushion. He assumed a deceleration distance of 1" at the leaf while the head and shoulders rotated forward one foot. In searching the literature for similar problems, Tom came across work done by the auto industry on shoulder strap forces during automobile accident impact. Their results, comparing analysis with experiments, showed that the forces observed in test were normally about 50% of those calculated. This led him to believe that his calculations were on the safe side and that a 1/4" slab of 7075-T6 aluminum would provide sufficient strength along with low inertial forces. There was no time for prototype tests before construction and drawings based on the aforementioned calculations were released.

As the pneumatic components began arriving, the first test was set up to determine barrier leaf actuation time. The system was timed from the solenoid valve electrical pulse to the final positioning of both barrier leaves. The actuation time was almost exactly .4 second, but unfortunately, the leaves did not open simultaneously. Tom had expected this problem because the regular-filter-valve package was located only a few inches from one leaf and across an aisle from the other leaf. He had a variety of orifices available to restrict the flow to the closer cylinder. By trial and error, he selected an orifice and found that it was the best size for both the Control Air and Emerson cylinders. Early results eliminated the Air Products cylinder because it had a slow stroke, its air cushion was difficult to adjust, and its long length made installation awkward.

The test had not been held up by orifice selection and the Control Air and Emerson cylinders continued toward their goal of 1.5 million cycles. On October 20, 1970 one of the Control Air cylinders failed after 1.2 million cycles while the Emerson cylinder was still performing well at 1.44 million cycles. When the Control Air cylinder was new, the interference between the teflon rings and the cylinder wall had measured .008". However, wear had reduced this interference at zero at certain points and blow-by occurred. The Emerson cylinder used rubber O-rings for piston seals and when examined after 1.5 million cycles, there was very little sign of wear. The endurance of the Emerson cylinder was off set, however, by its poor air cushion. Its air cushion stroke length was 3/16" and its stroke deceleration was quite inferior to that of the Control Air cylinder which had a 7/16" cushion stroke length. Tom had to choose between performance and maintenance costs. On October 26, he decided to go with the Control Air cylinder.

However, before this decision was released to purchasing, the P.I. salesman was able to convince Emerson to extend their cushion stroke length and he was able to convince Tom to give them another try. On November 2, a second set of Control Air cylinders was put on test using light oil rather than heavy lubrication and by November 9, two new Emerson cylinders with 1/4" cushion stroke lengths had been evaluated. Results showed that the Emerson air cushion was still difficult to adjust and that certain servicing problems existed. Control Air was still the preferred unit.

Another problem was exposed at the time of the first Control Air cylinder failure. As the unit was being removed, the plastic tubing bent and a crease or kink resulted. Product Test recommended that back-up tubing be located. Tom felt this was a good idea, but he decided to stay with the present tubing as his selection of pneumatic tubing was quite restricted and locating back-up tubing would take excessive time. Fortunately, this decision has proved reasonable as no other incidents of tube kinking have been reported.

On November 23, 1970 Tom finally released the Control Air cylinder for purchase although tests were continuing on their second set of cylinders and Emerson was still trying to improve their air cushion. A set of modified Emerson cylinders arrived on December 7. On December 23, the second set of Control Air cylinders experienced failure at 1.26 million cycles, again due to teflon ring wear. A decision was then made to look at a Control Air cylinder with rubber O-ring seals and to test the newly arrived Emerson cylinders. The new Control Air cylinders were purchased but there was never time to test them. The third generation

Emerson cylinder air cushion was still inferior to the Control Air, however, it was good enough to swing the maintenance cost versus performance balance in its favor. Also in its favor was the fact that Tom felt that it was a higher quality unit. On February 22, 1971, he released the Emerson cylinder as the replacement of the Control Air.

The remaining pneumatic components were also released for purchase by this time. The Success pressure regulator and the PVI solenoid valve had each accumulated 3.3 million cycles without any difficulties. The Success filter (Exhibit B-4, Figure 23) however had been replaced by a Johnson filter because of the failure of its automatic drain diaphragm. As air is removed from the filter bowl due to cylinder stroke, the pressure drop raises the diaphragm thus opening the drain seal. However, this diaphragm was made of neoprene rubber which warped when it became wet (Exhibit B-4, Figure 24) causing the diaphragm seal to leak thereby preventing drainage. Fortunately, there were still distributors visiting IBM trying to sell pneumatic products and Tom quickly located the Johnson unit which had a polycarbonate diaphragm and functioned properly.

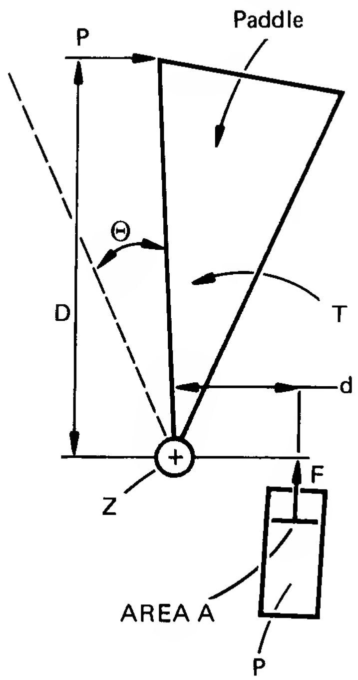
The filter and cylinder changes caused little packaging redesign as the filters were interchangeable and the Emerson cylinder required only a change in mount spacing. Consequently, when the Emerson cylinder was selected the tests were being run with the final pneumatic components in their final configuration.

The final barrier specification test was the structural strength test. Because of the crowded testing schedule, it was not run until April 1971.

A 200-lb sand bag was padded with foam, supported on an inclined track, and allowed to impact the gates at three miles per hour. After 500 such impacts, the gates were examined and no structural damage or variation in performance was observed. A vertical load of 150 lbs was then applied to each barrier (300 lbs total) and again, no damage occurred.

By this time, Tom was well into his next assignment of designing equipment for preparing the stations for unit installation. He could now look back at the barrier design with a great deal of satisfaction and some amazement. He said it was rather seldom that 90% of your components and design operate properly the first time around. By working closely with the Product Test group he was able to eliminate inferior components quickly and to develop a reliable barrier system within his time constraint. Tom did not feel he had a perfect system, however, the barrier system did work and as tests showed, it worked very well!

Barrier Calculations for Pressure Capacity



A minimum stall force P = 10 lb is required.

P = 10 lb D = 15" Θ = 30° d = 6"

$I = 2.05$ in-lb-sec^2 (for entire system)

Assuming the minimum force F to equal the stall force, we shall determine the time (t) which is required to drive paddle thru angle of 30°

$T = PD = I\alpha$

$T = 15 \times 10 = 2.05\alpha \quad \alpha = \frac{150}{2.05} = 73$ rad/s

$\Theta = 30 = \frac{\pi}{6}$ rad $\quad \Theta = \frac{1}{2}\alpha t^2 \qquad t^2 = \frac{2\Theta}{\alpha}$

$t = \sqrt{\frac{2\pi}{6 \times 73}} = \sqrt{1.43 \times 10^{-2}} = 1.2 \times 10^{-1}$ sec $= \underline{\underline{0.12 \text{ sec}}}$

$Fd = PD \quad F = \frac{10 \times 15}{6} = 25$ lb $\quad F = pA$

$A = 1.77$ in^2 for 1.5" dia. cylinder

$F = pA \quad p = \frac{25}{1.77} = 14.1$psi Allowing 50% for friction losses

$p = \underline{\underline{21.2}}$ psi

To time t = .12 sec we must add valve time & electrical delays & meet spec. of t = .400 ± .100 sec.

Schematic of Gate Pneumatics

Solid lines & arrows indicate air flow during closed (energised) mode.

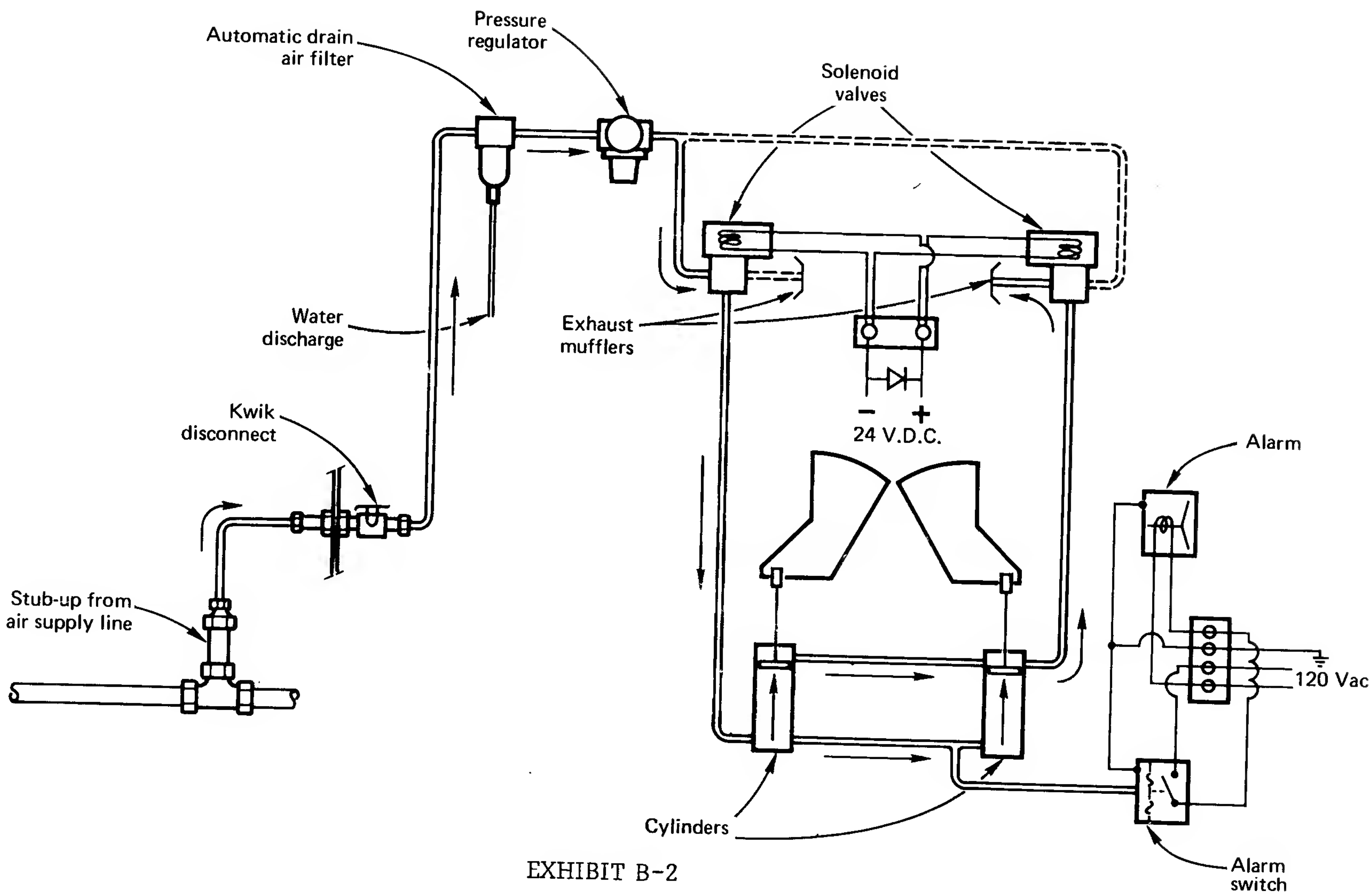


EXHIBIT B-2

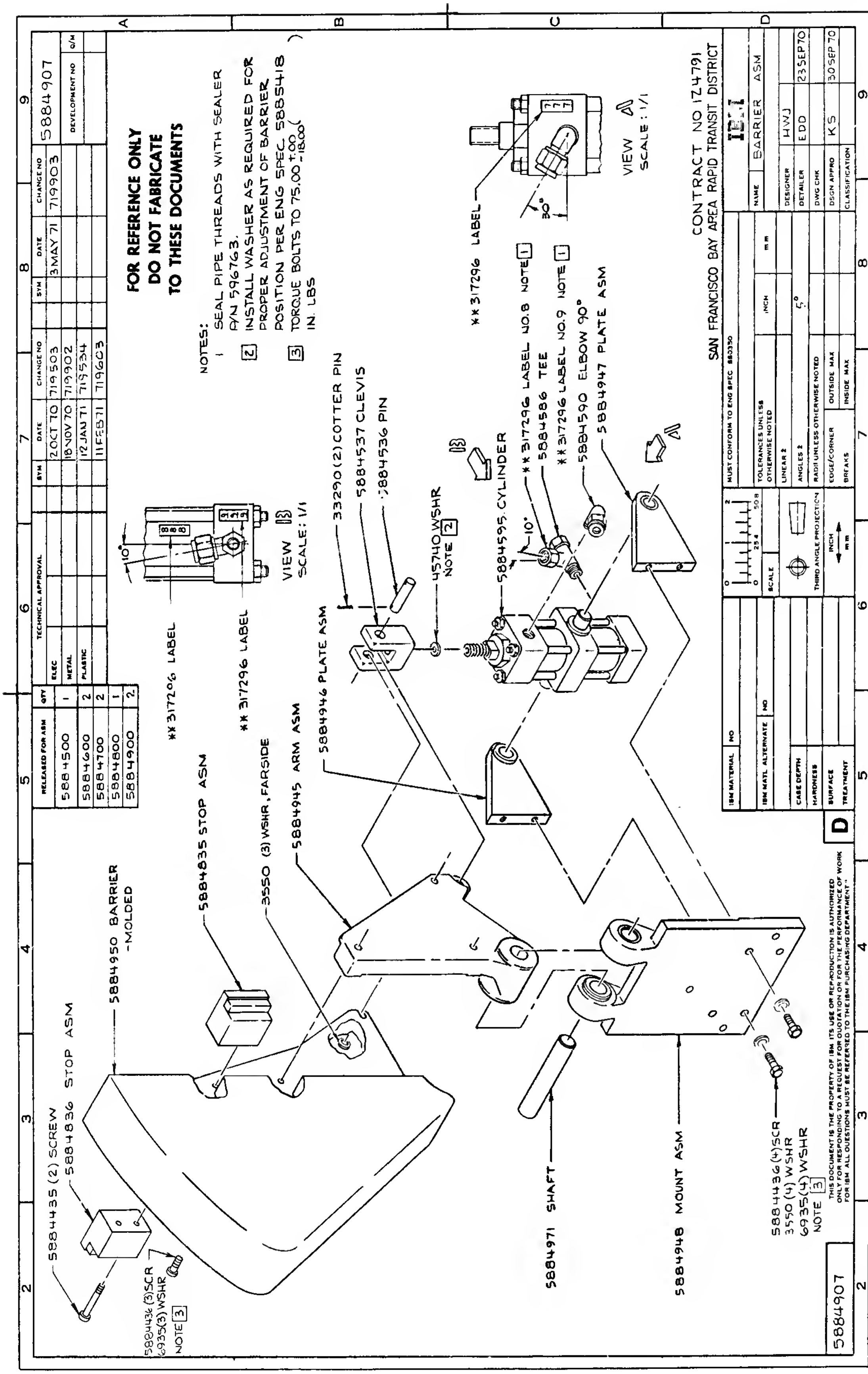

FOR REFERENCE ONLY
DO NOT FABRICATE
TO THESE DOCUMENTS
NOTES:
1 SEAL PIPE THREADS WITH SEALER P/N 596763.
2 INSTALL WASHER AS REQUIRED FOR PROPER ADJUSTMENT OF BARRIER POSITION PER ENG SPEC 5885418
3 TORQUE BOLTS TO 75.00 +.00 -18.00 IN. LBS
5884907
CONTRACT NO 1Z4791
SAN FRANCISCO BAY AREA RAPID TRANSIT DISTRICT
BARRIER ASM
5884435 (2) SCREW
5884836 STOP ASM
5884950 BARRIER -MOLDED
5884835 STOP ASM
3550 (3) WSHR, FARSIDE
5884945 ARM ASM
5884946 PLATE ASM
33290 (2) COTTER PIN
5884537 CLEVIS
5884536 PIN
45740 WSHR NOTE 2
5884595 CYLINDER
** 317296 LABEL NO.8 NOTE 1
5884586 TEE
** 317296 LABEL NO.9 NOTE 1
5884590 ELBOW 90°
5884947 PLATE ASM
** 317296 LABEL
VIEW A SCALE: 1/1
VIEW B SCALE: 1/1
5884971 SHAFT
5884948 MOUNT ASM
5884436 (4) SCR
3550 (4) WSHR
6935 (4) WSHR
NOTE 3
5884436 (3) SCR
6935 (3) WSHR
NOTE 3
RELEASED FOR ASM
5884500
5884600
5884700
5884800
5884900
20CT70 719503
18NOV70 719902
12JAN71 719534
11FEB71 719603
3MAY71 719903
DESIGNER HWJ
DETAILER EDD 23SEP70
DSGN APPRO KS 30SEP70

EXHIBIT B-3

Figure 23: View of Miniature Filter showing location of Diaphragm.

a) End View

b) Side View

Figure 24: Close-up of Diaphragm showing ripple pattern.

ECL 185C

THE DESIGN OF THE BART-AFC BARRIER DRIVE SYSTEM AT IBM (C)

THE DESIGN OF THE BART-AFC BARRIER DRIVE SYSTEM AT IBM (C)

Development of the Compressor System

When the decision was made in late June 1970 to adopt pneumatically operated barriers, the BART group at IBM had not developed a reliable compressor system. Steve Campbell recognized this as a critical area and he assigned Bob Everest the responsibility for compressor development in addition to his entire gate responsibilities. Tom McDowell was to help Bob with design support on the compressor while carrying full responsibility for the barrier. Bob started by examining the compressor system which had been built for the prototype barrier. This system, which was not felt to be sufficiently reliable and durable, used Thompson 175 psi compressors between 3/4 and 1 1/2 horsepower. Bob knew that he would need the same components, i.e., a compressor, a storage tank, filters, pressure switches, control circuitry, and a safety valve. However, he decided to start at the beginning with respect to sizing and the selection of manufacturers. He first had to make three major decisions: 1) What maximum pressure would he select? 2) What maximum flow rate was required? 3) Would he use a lubricated or oil-less compressor? The first decision reached was to use an oil-less compressor. The trade-offs associated with compressor type were between maintenance costs and replacement costs. The lubricated unit had a longer life than the oil-less unit, but it required regular oil changes. Bob felt that overall expenses would be minimized by going to an oil-less unit. This decision reduced perspective candidates to two. Only Thompson Manufacturing and Jennings, Inc. of the 12 major compressor manufacturers contacted made oil-less compressors in the 1-2 horsepower range.

The selection of an operating pressure for the compressors was also a trade-off decision. As space for the compressor system was limited,

Bob was committed to a ten gallon storage tank. Therefore, high pressure would increase the storage capacity of the system and the compressor idle time would be extended. This would allow extended cooling time for the compressors during rush hours and would allow all-night off-time in many stations. However, as the output pressure of a compressor increases, its efficiency decreases due to leakage losses. Bob's estimated 25% decrease in efficiency between 175 psi and 75 psi systems led him to favor the 75 psi compressor. But, more information concerning cylinder pressure and required air flow was necessary before compressor requirements could be completely specified.

Fortunately, this information was almost instantly available. First, the specification change reduced the barrier resistance force to 10 lbs (Part B) and second, expected passenger flow rates were provided by BART (last two pages of Exhibit C-4). Using a 21.2 psi cylinder pressure, the cylinder and line volumes and the assumed losses, Bob calculated that .045 SCF (cubic feet of air at 70°F and 14.7 psi) were required per passenger operation. Using one compressor unit per gate array he calculated that the heaviest passenger load predicted by BART required 6.4 SCFM (SCF per minute). These calculations led him to the conclusion that the flow of air through the compressor was a more critical factor than the pressure level. Therefore, because the 75 psi compressor has nearly double the flow rate (SCFM) of the 175 psi compressor of the same horsepower and because of its lower leakage losses, Bob decided to use the 75 psi compressor.

The final factor in determining compressor specifications was Bob's decision to use two compressors in each unit. He would size the com-

pressors such that their total output met the performance expectations. These units would provide the benefit of continued operation during failure of one compressor. A schematic of the compressor system is shown in Exhibit C-1 and part of the report leading to the selection of one horsepower compressors is shown in Exhibit C-3. Again, using the maximum usage station, Bob calculated that the compressor unit would operate at a 10% duty cycle per compressor. This led to the question of using smaller compressors at the lighter duty stations. He felt that the cost savings of a smaller compressor would be offset by the longer life of the 1-horsepower compressors and that the added expense of stocking multiple compressors and fixtures made the concept impractical. The cost of paperwork alone on maintaining a part number is quite high.

Bob now had to choose between the two 75 psi, one horsepower, oil-less compressors that were commercially available. The Thompson model 75-Z was a single ended compressor with three cylinders mounted on an electric motor. The Jennings model 732-40 was a double ended compressor with a pair of small cylinders mounted on each end of an electric motor. Of these two configurations, the Thompson was the more compact and therefore easier to package. The Thompson compressor also used an extra heavy duty motor and had a surge chamber to reduce output pressure fluctuations. However, Bob felt that it was necessary to test the Jennings compressor in case problems developed during the tests of the Thompson compressor.

The Product Test Group was notified of the upcoming compressor tests and were advised that they should include the usual safety, serviceability, function, thermal analysis, endurance and environmental tests,

and initial and final inspection. Tom McDowell was to assist Product Test in the design of necessary test enclosures and Jim Cochron was given responsibility for the electrical control system. The control system for these tests first turned on one compressor to pump up the storage tank. At 75 psi, a pressure switch shut down the first compressor and opened a valve to dump the storage tank. When the storage tank pressure was reduced to 45 psi, the tank valve was closed and the second compressor was turned on to initiate a new cycle. Jim decided to contract this circuit design to an outside vendor and to concentrate his efforts on the design of the production control system.

Jim's design however required a decision from the mechanical group. The compressors could be turned on alternately to pump up the storage tank with both turned on only when the load was beyond the capacity of a single compressor, or they could be used as primary and secondary units, with the secondary unit turned on only in heavy load situations. The latter system allows one compressor to be used in reserve, but it would be subject to storage type problems in light duty stations. Also, in a situation where pump up cycles occur very often the alternating system gives the compressor heads more time to cool off. The value of this cooling period was questioned, but it was felt to be doing some good and it was a factor in the decision to go to an alternating system.

Two Thompson compressor sets and one Jenning compressor set, each in a complete system configuration, were prepared for test. The complementary components were supplied by P.I. Again, this selection was based on the quality of the P.I. line of products and on the competence and availability of their salesman. Also, these were standard and relatively simple components and a thorough search for manu-

facturers appeared uneconomical. The first Thompson compressor system was placed on test on October 6, 1970, one month later than expected. These tests were to simulate one equivalent customer year which required 1752 hours per compressor set or 876 hours per compressor. However, after only 25 hours, both Thompson compressors had experienced valve failures. This was obviously disconcerting to the Engineering Group and they immediately contacted Thompson Manufacturing. They were assured that this was an unusual failure and were given replacement valves. The broken valves were sent to both Thompson and to the IBM Failure Analysis Laboratory for inspection. Both compressors were repaired and functioning by October 20. On November 2, two sets of Thompson compressors were running well, a set of Jennings compressors had started test, and Bob had received the reports on the original valve failures.

Thompson reported finding a sticky residue on the valve. They hypothesized that the valve may have momentarily held, increasing the cylinder pressure and stress in the valve. Then upon opening suddenly, high dynamic forces would also cause over-stressing. They concluded that the failure was due to overstress caused by contamination. The IBM Failure Analysis lab report identified the failure as fatigue, probably due to poor material or poor heat treatment. These conflicting reports did not encourage corrective action and Engineering decided to let the testing continue, watching closely for valve problems.

During the tests, the automatic tank drain and the line filter had to be replaced. The automatic drain was the same Success unit that had been used on the barrier system and it was also replaced by a Johnson unit. The 5-micron line filter between the storage tank and the barrier

became clogged with carbon dust after 400 hours. When ultrasonic cleaning failed to improve air flow, the filter was replaced with a 25-micron filter, and no further problems were observed. The Product Test Group also examined the safety, serviceability, and thermal characteristics of the compressor. The safety test concluded that the fan shroud on the Thompson compressor was insufficient to protect service personnel and a more extensive shroud would have to be used. Thompson Manufacturing was first asked to supply such a shroud but when they were unable to do so, Tom McDowell designed an add-on screen. Serviceability tests indicated that a bearing puller was necessary to remove the fan and therefore one was included in the tools required for cylinder ring maintenance. Numerous temperature and environmental tests were performed with plywood enclosures simulating the air flow and heat transfer characteristics of the final compressor packaging. These tests gave engineering a more complete understanding of the compressor's characteristics and assured them that the compressors would remain within their allowable temperature range and would function in the specified environments.

On December 14, after 19.8 equivalent customer weeks, the tests on the Jennings compressor set were terminated because their floor space was required for other experiments. At that time the two Thompson sets had accumulated 33.0 and 21.8 equivalent customer weeks respectively with no evidence of trouble. With packaging design establishing a strong commitment to the Thompson compressor, Bob advised against the expense of further Jennings compressor tests and all agreed. However, Bob was still somewhat concerned over the Thompson valve failures and decided to get a few more hours on the compressors before releasing them for

purchase. Finally, when a change of ownership at Jennings indicated long delivery delays and the Thompson compressor had successfully completed 44 equivalent customer weeks, Bob released the Thompson compressor as well as all of the compressor system components. This was on January 15, 1971, just nine months before the scheduled delivery of complete AFC systems.

On February 4, 1971, the first set of Thompson compressors had completed 52 equivalent customer weeks and was dissambled for inspection. Product Test found numerous problems including cracks in two pistons, a broken output check valve, a broken centrifugal pressure unloader arm, and grease and carbon buildup on connecting rod bearings. Except for the piston cracks, these failures were associated with secondary components and in their present usage performance was not degraded. The pistons were eventually put back into the compressors and after hundreds of additional hours with no failures they were not considered a problem area. The valves, interestingly enough, were found to be in excellent condition.

In addition to looking for failures, Bob was concerned with output degradation. Product Test found that the data published by the manufacturer was applicable only during the first minute of operation when the compressor was cool. As the compressor operating time increased the temperature of the heads increased and the output flow decreased. Therefore Product Test ran output flow versus power on hours after the compressor had been operating for 20 minutes. An envelope of the results for the four Thompson compressors is shown in Exhibit C-2. This also brings up the question valve and piston life. Thompson quotes a valve

life of 2,000 hours and a ring life of 4,000 hours or 2.3 and 4.6 equivalent customer years. However, they do not indicate the performance degradation of the compressor relative to these life times. Therefore, for the AFC system performance degradation may cause the compressor to fail functionally long before a ring or a valve fails.

The results of Exhibit C-2 conclude that two compressors on the worst-case curve would require maintenance after 400 hours or approximately six months in order to meet the worst case one minute requirement of 5.2 SCFM (3.1 SCFM per compressor). However, these curves are plotted at 78 psi while the compressor would actually operate at a lower average pressure.

The second set of Thompson compressors was not disassembled after its one equivalent customer year but was allowed to continue running for a total of 1500 hours. Inspection showed that the compressors were still functioning correctly, but that their output was reduced due to continued wear. While the reduced volume would not be sufficient for a heavy duty station, it would be quite adequate for many low duty stations. This led Bob to establish separate maintenance and replacement requirements for heavy and light duty stations (Exhibit C-4).

On March 15 the barrier and compressor system began prototype tests. These tests are carried out on complete systems which are generally constructed by engineering rather than manufacturing. This was the first operation of the compressor system with Jim Cochron's control box. The tests examined all functions and simulated all environmental and worst case inputs. The system was also operated by specially

designed robots and the pneumatic system underwent approximately 10 equivalent customer months with no functional problems. In order to facilitate installation and repair, some of the pneumatic fittings were changed from screw fittings to quick disconnects.

The Prototype tests were completed on June 29 and the Production tests were started immediately. These tests take equipment as it would come off the assembly line, examine all adjustable settings and test all system functions. The equipment was subjected to worst case voltage, line frequency and environmental situations. During each night the robots were set up and accumulated a total of 800,000 simulated customer operations.

With various tests accumulating over two years of equivalent customer operation without the occurrence of any serious problem, the engineering group was confident that they had developed a functional and reliable barrier system. On September 3, 1971 the first AFC systems were ready to leave IBM for installation at BART stations.

Schematic of Compressor System

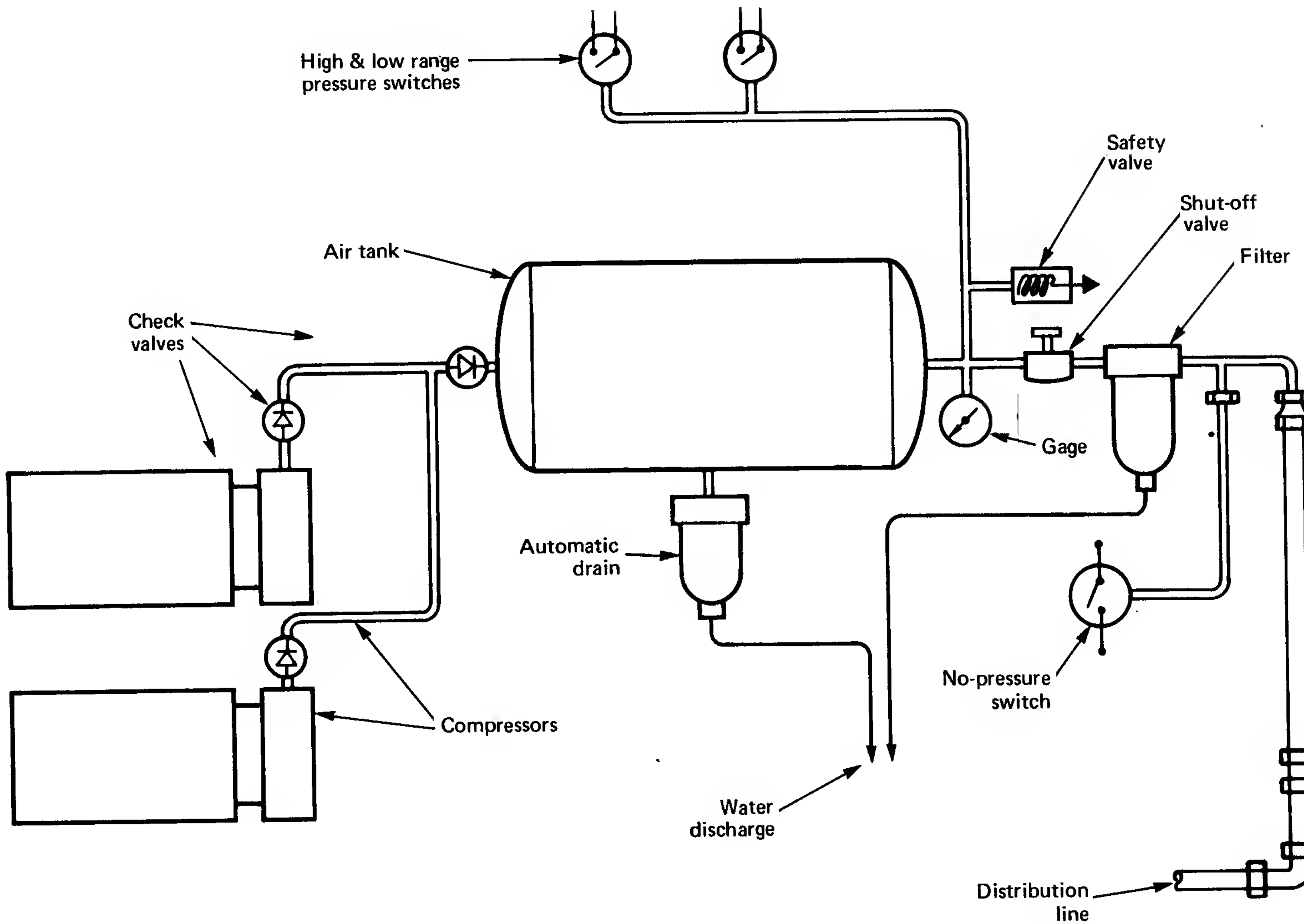


EXHIBIT C-1

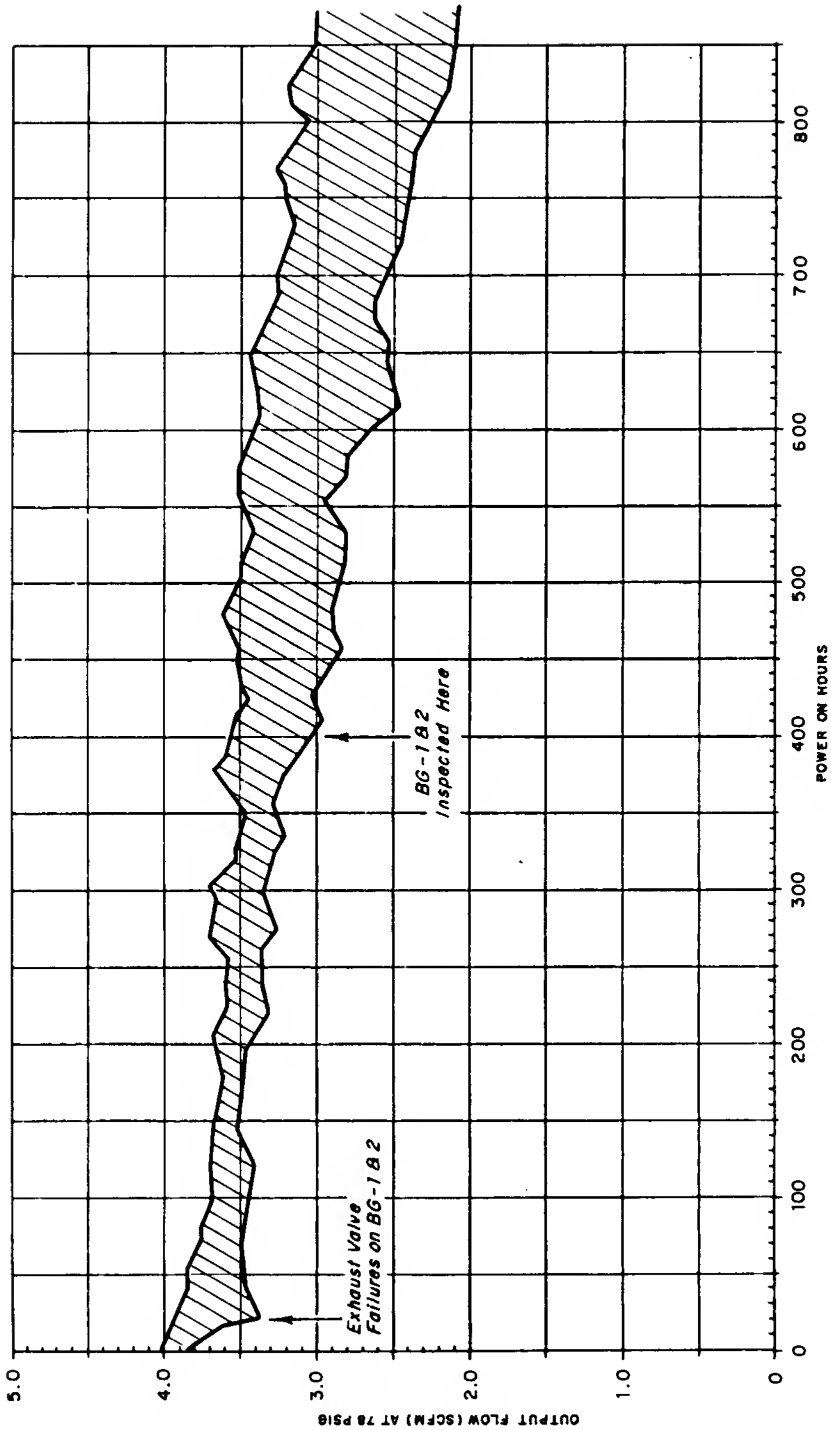


FIGURE 19 - ENVELOPE OF OUTPUT FLOW VS. POWER ON TIME FOR BG-1,2,3,4.

EXHIBIT C-2

Pneumatic Data for BART System

The following figures show the cubic feet per minute of free air (CFM) required to operate the barrier system based on the peak traffic load estimates (see attachment) at the busiest station. These loads will be compared to the compressor system capacity to determine the system operating characteristics.

Barrier Characteristics

One barrier operation which represents one patron passing through requires:	.045F
Maximum number of patrons @ Montgomery Street station per attached data sheet is:	
Peak 1 min. load	142
Peak 5 min. load	568
in (CFM)	
Peak 1 min. load	6.4 CF
Peak 5 min. load	25.5 CF

This peak load is estimated to occur at 3 arrays of the Montgomery Street station and the 1 array at Fruitvale. For comparison, the next highest patron load is at the Balboa Park station, where the peak loads in patrons are:

Peak 1 min. load	79
Peak 5 min. load	315

which are less than 60% of the forementioned loads of Montgomery and Balboa Park. The figures for the Coliseum exceed the capacity of the gates and require additional evaluation to determine if these figures are representative of actual operation.

EXHIBIT C-3

Compressor System Characteristics

The compressor system, which includes two compressors, one storage tank, fittings, and controls to cycle the pumps alternately as required, is set up to normally operate on one pump. Should the load exceed the capacity of a single pump, the additional pump is turned on to bring the system pressure up to the level where the system automatically shuts off. A single compressor has the following capacity figures which allow for an approximate 15% loss from rated output:

One Compressor	@75 PSI	@ 55 PSI
CFM Output	4.0	4.42
Patrons/Min.	90	98
Patrons/5 Min.	450	490

To handle the peak load previously specified for the Montgomery and Fruitvale stations, the second compressor would normally be turned "on" which gives the output of:

Two Compressors	@ 75 PSI	@ 55 PSI
CFM Output	8.0	8.84
Patrons/Min.	180	196
Patrons/5 Min.	900	980

The output of the compressor set exceeds the maximum requirements of the Montgomery and Fruitvale stations by a comfortable margin.

Storage Capacity

The storage tank in the compressor system which is a ten gallon coded ASME tank will operate the barrier 48 times (48 patron passages) when the tank is discharged from 75 PSI to 55 PSI. During the peak load period at Montgomery, the tank discharge will occur in approximately 0.3 sec. and pump up with two pumps would occur in approximately one minute. The cycling rate under heavy load periods is too high, and therefore the installations at Montgomery, Fruitvale and any other stations requiring a two-compressor operation should be operated on a timed cycle, and possibly have compressor systems interconnected where several systems are used.

EXHIBIT C-3

Date:	August 10, 1971
From (location or U.S. mail address):	FTS Mechanical Development SDD San Jose
Dept. & Bldg:	H26/060
Tieline & Tel. Ext.:	1374

ECL 185C

IBM

Subject: Suggested Compressor Replacement Point; and Analysis of Replacement Criterion

Reference: BART-AFC Pneumatics Components Development Test; Product Test Report for Project No. 6705

To: File

The purpose of this memo is

1. To discuss the factors used in calculating the duty cycle;

2. To show method of calculating worst case duty cycle of seven aisle array at Montgomery Station;

3. To explain proposed method for determining time when compressor should be replaced.

<u>Factors Used in Calculation of the Duty Cycle</u>

The attached charts prepared by Mr. Paul Kirby were used to determine loadings at the different stations in the system. The peak minute loads were used to calculate the maximum flow requirements; however, the average flow of patrons was used to determine the duty cycle. Included in the following calculations is an allowance for the effect of banking in the gate operations. Conservative figures for banking are during rush hours (one hour morning and evening); the "Entry" operation will bank (no barrier movement) for 50% of patrons passing through; "Exit" operation will bank for 25% of patrons passing through. (Per discussion with Mr. Paul Kirby 8/3/71

EXHIBIT C-4

Duty Cycle Calculation

The following calculations are for the Montgomery Street Station which carries the heaviest traffic per the attached charts:

The peak hour traffic is 18,617 patrons representing 18.6% of the day's load. Per day the load is 18,617 ÷ .186 = 100,000 patrons per day using 27 aisles in 6 arrays. The average load per aisle is 100,000 ÷ 27 = 3700 patrons per aisle per day. The seven aisle array which is driven by one compressor set has an average maximum load of 7 x 3700 = 25,900 patrons per day. This figure will be reduced by the tickets which are banked during gate operation. Taking 7/27 of the maximum hour for the seven aisle array we have 18,617 x 7/27 = 4825 patrons/hour.

With banking:

1 hour @ .50 (assuming "Entry")

.5 x 4825 = 2412 patrons

1 hour @ .75 (assuming "Exit")

.75 x 4825 = 3616 patrons

EXHIBIT C-4

Remaining patrons for 22 hours are:

25900 - (2 x 4825) = 16,250 patrons

These three figures added represent total corrected gate operation per day:

16,250
2,412
3,616

22,278 gate operations

The barrier operation requires .045 standard cubic feet of air per patron operation, and total air use per day becomes

22,278 x .045 = 1010 cubic feet free air per day

The duty cycle may now be determined by comparing the compressor set output per day with the daily requirements for the passenger load.

The average compressor set output at 75 PSI at approximately 400 hours of operation was 3.2 SCFM per compressor. During the pump-up cycle the efficiency varies to give an average of 3.5 SCFM per compressor, which will be used to calculate the compressor set output which is:

3.5 SCFM x 60 min. x 24 hrs. x 2 units =

10,100 cubic ft/day and

$$\% \text{ Duty Cycle} = \frac{\text{Air used/day} \times 100}{\text{Compressor set output/day}} = \frac{1010}{10100} \times 100 = 10\%$$

This maximum duty cycle figure for the Montgomery Street Station is over 10 times as great as the duty cycle of some lesser used stations such as Orinda and Lafayette.

Compressor Replacement Criteria

With the wide variation in duty cycle and maximum load requirements, the criteria for the replacement of compressor should be carefully evaluated as field experience is gained, so that unnecessary compressor replacement

is avoided. The suggested criteria is to designate an A & B replacement schedule based on compresor output when checked during PM periods. The A schedule, which is for the stations so indicated on the attached sheets, requires a minimum output of 5 SCFM @ 50 PSIG per compressor set. (Handles approximately 550 patrons per 5 minute period with an extra allowance of 48 patrons from the pressure tank discharge and a reserve of .5 SCFM (50 patrons per 5 minute period) if load causes pressure to drop to 30 PSIG.

The B schedule, which is for the less-used stations, calls for a 3 SCFM output at 50 PSIG per compressor set. These schedules will be included in the maintenance manuals as criteria for compressor replacement.

Tom McDowell

STATION	Number of arrays	Peak array total exit & entry per station (1)	Peak array % of daily total (1)	Peak 5 minutes (station) (1)	Peak 5 minutes per array (2)	Peak 1 minute per station (3)	Peak 1 minute per array (4)	Number of aisles per array	Compressor Schedule
North Berkeley	2	640	17.6	80	#1-26 #2-54	20	#1-6 #2-14	3 3	B
El Cerrito Plaza	1	632	17.2	84	84	21	21	3	B
El Cerrito Del Norte	2	835	17.9	111	#1-37 #2-74	28	#1-9 #2-19	3 3	B
Richmond	1	481	14.7	72	72	18	18	3	B
Rockridge	1	666	17	85	85	22	22	3	B
Orinda	1	294	22.9	30	30	8	8	3	B
Lafayette	2	374	22.5	38	#1-12 #2-26	10	#1-3 #2-7	3 3	B
Walnut Creek	1	694	22.4	71	71	18	18	3	B
Pleasant Hill	1	551	22.7	54	54	14	14	3	B
Concord	1	713	20	79	79	20	20	3	B
Lake Merritt	2	2,811	17	463	#1-154 #2-309	116	#1-39 #2-77	3 3	B
Fruitvale	1	4,134	20.3	550	550	138	138	6	A
Coliseum	1	1,381	20	282	282	71	71	8	A
San Leandro	1	961	20.1	266	266	67	67	4	B
Bay Fair	1	803	19.7	109	109	28	28	3	B
Hayward	2	1,438	19.1	193	#1-64 #2-129	49	#1-16 #2-33	3 3	B
South Hayward	1	422	21	59	59	15	15	3	B
Union City	1	780	23	109	109	28	28	3	B
Fremont	1	853	24	122	122	31	31	3	B
Coliseum Sport Night	1	20,000 (5)	--	2400 (6)	The loading will exceed gate capability . With 10% min. Fares, patron flow=35 pct/min/ccc. This flow rate should result in 30 cycles of barrier operation/aisle.			7 effective aisles one direction	A

(1) = From BART traffic analysis
(2) = Assumes worst case distribution of 2:1 ratio for arrays
(3) = 25% of peak 5 minutes (station)
(4) = 25% of peak 5 minutes per array
(5) = PKirby estimate
(6) = 12% of peak 1 hr.
NOT SPORT NIGHT TRAFFIC

EXHIBIT C-4

STATION	Number of arrays	Peak array total exit & entry per station (1)	Peak array % of daily total (1)	Peak 5 minutes (station (1)	Peak 5 minutes per array (2)	Peak 1 minute per station (3)	Peak 1 minute per array (4)	Number of aisles per array	Compressor Schedule
Daly City	1	1,920	15	206	206	51	51	3	B
Balboa Park	1	2,993	15	315	315	79	79	4	A
Glen Park	1	2,840	15	299	299	75	75	4	A
24th St. Mis.	1	2,818	15	293	293	74	74	5	A
16th St. Mis.	1	2,734	15.2	269	269	69	69	4	A
Civic Center	3	3,680	16.3	452	#1-113 #2-226 #3-113	113	#1-28 #2-57 #3-28	3 3 3	B
Powell Street	6	3,732	16.5	455	#1-50 #2-101 #3-51 #4-101 #5-51 #6-101	114	#1-12 #2-26 #3-12 #4-26 #5-12 #6-26	3 3 3 3 3 3	B
Montgomery	6	18,617	18.6	2,559	#1-284 #2-568 #3-284 #4-568 #5-284 #6-568	640	#1-71 #2-143 #3-71 #4-142 #5-71 #6-142	3 5 3 6 3 7	A
Oakland West	1	693	16	84	84	21	21	3	B
12th Street	3	2,431	16.5	291	#1-73 #2-146 #3-72	73	#1-18 #2-37 #3-18	3 3 3	B
19th Street	3	1,904	15.8	237	#1-59 #2-119 #3-59	60	#1-15 #2-30 #3-15	3 3 3	B
MacArthur	1	988	16.3	121	121	31	31	3	B
Ashby	1	820	17.2	107	107	26	26	3	B
Berkeley	4	2,228	16.2	302	#1-50 #2-101 #3-50 #4-101	76	#1-13 #2-25 #3-13 #4-25	3 3 3 3	B

(1) = From BART traffic analysis
(2) = Assumes worst case distribution of 2:1 ratio for arrays
(3) = 25% of peak 5 minutes (station)
(4) = 25% of peak 5 minutes per array

EXHIBIT C-4